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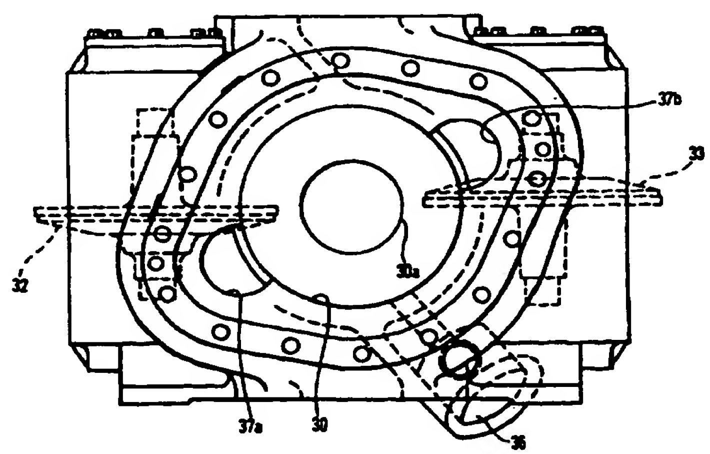

(57) Abstract

A single screw compressor having at least two pumping chambers (10A, 10B) of variable volume formed in the flutes of a screw, each of which receives fluid to be compressed from a primary inlet (14:34) when the respective pumping chamber has a first volume at least close to its maximum volume and each of which discharges compressed fluid from an outlet (13a, 13b) when the respective pumping chamber has a second volume which is less than said first volume, ne f said pumping chambers (10B) having a secondary inlet (12b, 36) open to the said one chamber (10B) when it has a third volume intermediate the first and second volumes after said one chamber (10B) is closed off from its primary inlet but before the said one chamber (10B) opens to its outlet, is characterised in that means (38a, 38b: 40) is provided on the compressor to connect the outlet of the other pumping chamber (10A) to the secondary inlet (12b, 36) of said one pumping chamber (10B).

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IMPROVEMENTS IN AND RELATING TO SINGLE SCREW COMPRESSORS

Technical Field

This invention relates to a single screw compressor having at least two pumping chambers of variable volume formed in the flutes of a screw, each of which receives fluid to be compressed from a primary inlet when the respective pumping chamber has a first volume at least close to its maximum volume and each of which discharges compressed fluid from an outlet when the respective pumping chamber has a second volume which is less than said first volume, one of said pumping chambers having a secondary inlet open to the said one chamber when it has a third volume intermediate the first and second volumes after said one chamber is closed off from its primary inlet but before the said one chamber opens to its outlet. Throughout this specification such a compressor will be referred to as "a single screw compressor of the kind specified".

The invention seeks to improve the efficiency of a single screw compressor (in particular a refrigerator compressor) of the kind specified.

Discussion of Prior Art

Single screw compressors are widely used in industry and features of them are described inter alia in US-A-3133695, US-A-3181296, US-A-3180565, US-A-3551082, US-A-3632239 and US-A-3752606.

It is known to improve the efficiency of a compression refrigeration system by using a single screw compressor of the kind specified and connecting the secondary inlet to a source of pressurised refrigerant gas at a pressure higher than that of the suction pressure in the primary inlet.

Compressor efficiency can be enhanced if refrigerant gas is also allowed to enter the system via an intermediate port (sometimes called herein an "economiser port"). As the economiser port is open to an intermediate stage of the compression process, for gas to enter the economiser port it must be at a higher pressure than the suction pressure. This is traditionally achieved by the use of an economiser vessel/heat exchanger and its associated control valves and pipework.

The operating principle behind the use of an economiser vessel/heat exchanger (sometimes referred to as "superfeed") is that liquid refrigerant is boiled off in the economiser vessel or heat exchanger at a pressure lower than the delivery pressure but higher than the suction pressure. The gas evolved from the vessel or heat exchanger is returned to the economiser port of the compressor whilst the liquid subcooled in the economiser vessel is led to the suction evaporator. This liquid subcooling increases the refrigeration capacity of the system. In addition, as the vapour evolved in the economiser vessel or heat exchanger is returned to the compressor at an intermediate pressure, less power is required to compress this vapour than if it were returned to the compressor at the suction pressure, thus effecting further economies.

Summary of the Invention

According to this invention a single screw compressor of the kind specified is characterised in that means is provided on the compressor to connect the outlet of the other pumping chamber to the secondary inlet of said one pumping chamber.

In one aspect, the invention can be seen as a way of modifying a prior art single screw refrigeration compressor having an economiser port to improve performance without the need for any economiser vessel or system-derived side load.

The secondary inlet need not be in a fixed position in the said one pumping chamber and both pumping chambers in a compressor of the kind specified can have such secondary inlets. Means can be provided to utilise conventional oil or liquid refrigerant injection in either or both pumping chambers.

Use of a compressor according to this invention may be further refined by employing an economiser facility (e.g. with a conventional economiser vessel or heat exchanger). Greater performance can be achieved in this way.

__Brief Description of the Drawings__

The invention will now be further described, by way of example, with reference to the accompany drawings, in which:

Figure 1 shows, schematically, a single-screw refrigeration compressor of the kind specified operating in conventional economiser manner,

Figure 2 shows the components of the schematic screw compressor of Figure 1 connected in accordance with the invention without an economiser vessel,

Figure 3 shows the components of the schematic screw compressor of Figure 1 connected in accordance with the invention but with an economiser vessel connected to one of the pumping chambers,

Figure 4 shows the components of the schematic screw compressor of Figure 3 further modified to utilise two economiser vessels connected in series,

Figures 5 to 8 are graphs showing respectively pressure/enthalpy diagrams for the compressors shown in Figures 1 to 4,

Figures 9, 10 and 11 show less schematically a single rotor, twin star wheel compressor, according to the invention, Figure 10 being a section on line X-X of Figure 9 and Figure 11 being a partially dismantled end view, and

Figure 12 is a bar chart comparing C.O.P. and duty values of a range of different single screw compressor arrangements.

Description of Preferred Embodiments

Although this invention relates to a single screw compressor, Figures 1 to 4 illustrate piston-in-cylinder pumping chambers for easier understanding of the inventive concept.

Figure 1 shows a prior art refrigeration compressor unit having two pumping chambers 10A and 10B each having a primary inlet 11a, 11b, a secondary inlet (or economiser port) 12a, 12b and an outlet 13a, 13b. In conventional manner, the two primary inlets communicate with a common suction connection 14 leading to the evaporator 16 of the unit and the two outlets communicate with a common high pressure delivery port 15 leading to the upstream end of the condenser 17 of the system. The expansion valve is shown at 18. For economiser operation the intermediate ports 12a and 12b are connected to a vapour outlet 19a of an economiser vessel 19.

Figure 2 shows how the compressor of Figure 1 is adapted in accordance with the invention. The outlet 13a of chamber 10A is now connected to the intermediate inlet 12b of chamber 10B. Both inlets 11a and 11b remain connected to a common suction line 14 but only outlet 13b now feeds the delivery port 15. In its simplest form the intermediate port 12a is not used and this is the arrangement shown in Figure 2.

Figure 3 shows a further modification with the addition of an economiser vessel 19. The vapour outlet 19a from this economiser vessel is returned to the intermediate port 12a.

Figure 4 shows a still further modification with two economiser vessels 19A, 19B connected. The vapour outlet from the first economiser vessel 19A, operating at the higher saturation pressure, is returned to the intermediate port 12b, where it mixes with the flow from the discharge port 13a as it enters chamber 10B. The vapour outlet from the second economiser vessel 19B is returned to the intermediate port 12a and flows into chamber 10A.

Unloading of the compressors 10A, 10B shown schematically in Figures 2, 3 and 4 can be undertaken in the normal way except that it is anticipated that the compression process in chamber 10A, pumping into the economiser port 12b, would be unloaded before the compression process in chamber 10B.

The pressure enthalpy diagrams shown in Figures 5 to 8 represent various refrigeration cycles starting with standard economised in Figure 5 and continuing with systems incorporating turbo charging shown in Figures 6, 7 and 8.

Figure 5 shows the pressure enthalpy diagram for a standard economised single screw compressor machine. Starting at point 1, compression of a non-economised machine would normally progress along line 1-3, but when economiser flow is introduced at an intermediate pressure Pe then, provided the temperature of this gas from the economiser vessel is lower than the temperature of the gas in the flute of the screw at this point then the temperature within the compressor will fall to point 4, compression will then continue to point 5 to achieve the delivery pressure Pd. Desuperheating and condensing of the gas will then take place as shown by line to 5 to 6. In a non-economised system the liquid would pass through an expansion device where the pressure would be reduced to the suction pressure Ps at point 10. However in an economised system some liquid is dropped in pressure to point 7 at the economiser pressure Pe where it boils off and cools the remaining liquid to point 8. Figure 5 is drawn in

respect of an open flooded economiser vessel but it is equally possible to accomplish this cooling in an indirect heat exchanger where the liquid that is cooled is retained at the higher condensing pressure. Finally the remaining liquid in the economiser is fed, via an expansion valve, to the evaporator, where it evaporates thus providing the required cooling and is ultimately returned to the compressor suction at point 1 via point 9. Thus the beneficial effect of economising can be seen by the increased enthalpy change from 9 to 1 in the economised mode when compared to the enthalpy change from 10 to 1 as applies to a non-economised arrangement.

Figure 6 shows the pressure enthalpy diagram for a simple single screw compressor in accordance with the invention. Starting at point 1, in a conventional system the compression process would normally progress along dashed line 1-4. However if the compression process is divided into two compression chambers such that the first compression chamber compresses over the entire required pressure range from suction pressure to delivery pressure, whereas the second compression chamber compresses over a reduced compression ratio (and thus with higher efficiency and reduced temperature) from suction pressure Ps to the pressure Pe at the economiser port of the first chamber, as shown by line 1-3. The gas from this second chamber is fed, via the economiser port of the first chamber, into the first chamber, where the two compression gas streams are combined. The compression process in the first chamber progresses along line 1-2, which is less efficient than that achieved in the second chamber due to the gas leakage effects from the higher pressure ultimately achieved in this compression chamber. The combined gas streams will then be further compressed in the first compression chamber to the final delivery pressure Pd at point 5. The cycle continues with desuperheating and condensing taking place between 5 and 6, the pressure then being dropped from 6 to 7 and the liquid boiled off in the evaporator from 7 to 1. In this instance the benefits are not

shown by an increase in enthalpy change in the evaporator, but rather by an increase in mass flow through the evaporator, together with an improvement in coefficient of performance (C.O.P.), which can not be represented in Figure 6.

Figure 7 shows the result of the addition of economising of the low compression ratio chamber. In this instance the two gas flows are combined in the first compression chamber from points 2 and 3, like the non-economised arrangement discussed above, and compressed together from pressure Pe to the final delivery pressure Pd at point 5. However the second compression chamber combines the economising effect described previously. Thus desuperheating and condensing of the gas takes place between 5 and 6. The liquid is then dropped in pressure to point 7 (pressure Pel) in the economiser vessel, where some liquid is boiled off and returned to the economiser port of the second compression chamber. This evaporation cools the remaining liquid in the economiser to point 8. Finally the remaining liquid in the economiser is fed to the evaporator, where it evaporates, thus providing the required cooling and is ultimately returned at pressure Ps from point 9 to the compressor suction at point 1. Figure 7 again shows an open-flooded economiser vessel but it is equally possible to accomplish this economiser liquid cooling in an indirect heat exchanger where the liquid that is cooled is retained at the higher condensing pressure. In this instance the beneficial effect of economising can be seen by the increase in enthalpy change in the evaporator shown by line 9-1 in this diagram when compared with line 7-1 of Figure 6.

Figure 8 shows the performance of a compressor similar to that used for Figure 7, but with the addition of means for providing economising to the first compression chamber. This compressor operates similarly to that previously described (for Figure 7) except that after the gas from the second compression chamber has been fed to the first compression chamber, further economising is introduced in one of two possible ways. Either additional high pressure economiser

flow is also introduced at the pressure Pe where the two compression chambers are combined, from an additional economiser vessel, or alternatively economiser flow from the second high pressure economiser operating at pressure Pem is introduced at a later stage either by pressure regulation or an additional later economiser port into the first compression chamber. In this instance the beneficial effect of additional economising can be seen by the increase in enthalpy change in the evaporator shown by line 12-1 in this diagram when compared with 9-1 of Figure 7.

In all four cases shown in Figures 5 to 8, the beneficial effect on power can be seen by either a reduction in the enthalpy at the end of the compression process or by a reduced pressure difference over which any gas flow is required to be compressed.

Figures 9 to 11 show a single main rotor, twin star wheel screw compressor externally of generally conventional design, but internally adapted in accordance with the invention.

Full details of the construction and operation of the compressor shown in Figures 9 to 11 are not included here in view of their familiarity to those skilled in the art but reference is made to the US patent specifications referred to previously, the entire contents of each of which is incorporated herein by way of reference.

Referring to Figure 9, this shows a plan view of the casing 31 of a single screw compressor in accordance with the invention. A single screw in the casing 31 provides flutes which serve as pumping chambers with star wheels 32 and 33. The flute terminating in a tooth of star wheel 32 on side 1 of the casing 31 serves as the chamber 10B in Figures 2, 3 or 4 and the flute terminating in a tooth of star wheel 33 on side 2 serves as the chamber 10A in those three Figures.

Figure 10 is a section on line X-X of Figure 9 and Figure

11 is an end view of the compressor of Figure 9 with the suction end cap removed. The screw is not shown in Figures 9 or 10 but occupies the bore 30 and is driven by the shaft 30a shown in Figure 9. The suction port 34 communicates with the low pressure end of each pumping chamber and in a conventional compressor both chambers will discharge to a common delivery port 35.

In a modified screw compressor according to the invention, the suction port 34 remains open to both sides of the compression process but the delivery ports from each side of the compression process are now separated. Side 1 of the compressor is piped up externally in the conventional fashion with the delivery connected to the condenser. However delivery gas from side 2 of the compressor is taken, either through an internal chamber or via an external pipe, to an economiser port 36 (not shown in Figure 9) provided on side 1 (the conventionally connected side).

Figure 10 shows the bore 30 for the screw and recesses 37a, 37b to receive the sliding valve gear conventional in a single-screw compressor. It also shows partitions 38a, 38b which separate the delivery ports from the two sides of the compressor, allowing the required separation of the deliveries from the two sides of the compressor.

Figure 11 is an end view of the compressor shown in Figure 9 in the direction of the arrow XI, with the delivery end cover 39 removed. One economiser port 36 is shown which is connected by an internal passage 40 to the delivery of side 2 of the compressor. To accommodate the extra volume of gas flowing into the economiser port 36 in a compressor according to the invention, this port can be enlarged compared to the economiser port provided in a conventional machine.

If required, the economiser connection of side 2 (shown at 41 in Figure 9) may be used to provide a conventional economiser facility.

Unloading of a single screw compressor modified in this way can be accomplished by unloading the compression processes on both sides together in the normal way. However, for efficient part-load operation the compression process pumping into the economiser port 36 should be unloaded first, followed by unloading of the remaining compression process.

The economiser port 36 need not be in a fixed position in the compressor casing, but may be capable of moving within the casing to maintain an optimum part-load economiser position.

Main oil injection may be into either both sides or may be into one side only dependent upon cooling and sealing requirements.

Liquid injection cooling may be used with injection into either side or the flow from delivery to economiser or any combination of these positions.

The introduction of a non-return valve between the discharge ports 13a and 13b, together with an automatic or manual isolating valve between 13a and 12b would permit changeover from conventional operation to the supercharged operation to which this invention relates.

Extracts from the results of tests conducted are shown in graphical format in the bar chart of Figure 12.

Figure 12 shows the increase in performance over and above that secured by a standard single screw compressor connected in the traditional fashion. Both coefficient of performance (C.O.P.) and duty variation are shown.

The base line for the graph of Figure 12 (100% performance compared with the standard machine) indicates the actual performance achieved with single screw test compressors connected to operate in the conventional format.

The first bar in each section in Figure 12 indicates the initial test result with the compressor modified in accordance with the invention. The second bar in each section indicates the performance achieved when oil injection rates are optimised for a compressor in accordance with the invention. The third bar in each section indicates the effect of adding an economiser system to the unused economiser port of the low pressure compressor. The fourth and fifth bars in each section are a repeat of the tests shown in the second and third bars, but with one of the original compressors replaced by another similar unit. These latter repeat tests were undertaken to ensure that the gains in performance recorded were not peculiar to the actual compressors tested.

The pressure of fluid fed to the secondary inlet or economiser port will normally be in the range 30% to 70T of the final discharge pressure from the supercharged compressor. Typical figures employed in practice lie between about 50% and 65%.

The tests were conducted on Hallscrew 2116 compressors and Figure 12 is an average of several test results.

A co-pending International application of even date entitled "Improved Compressor Arrangement and method of working the same" covers compressors of any suitable type interconnected as shown in Figures 2 to 4 and methods of operating such compressors to enjoy the advantages of Figures 6 to 8.

Claims

1. A single screw compressor having at least two pumping chambers (10A, 10B) of variable volume formed in the flutes of a screw, each of which receives fluid to be compressed from a primary inlet (14:34) when the respective pumping chamber has a first volume at least close to its maximum volume and each of which discharges compressed fluid from an outlet (13a, 13b) when the respective pumping chamber has a second volume which is less than said first volume, one of said pumping chambers (10B) having a secondary inlet (12b, 36) open to the said one chamber (10B) when it has a third volume intermediate the first and second volumes after said one chamber (10B) is closed off from its primary inlet but before the said one chamber (10B) opens to its outlet, **characterised in that** means (38a, 38b: 40) is provided on the compressor to connect the outlet of the other pumping chamber (10A) to the secondary inlet (12b, 36) of said one pumping chamber (10B).

2. A single screw compressor according to claim 1, **characterised in that** the bore (30) containing the single screw is surrounded by two delivery outlets (13a, 13b) formed in the casing (31) of the compressor, these delivery outlets (13a, 13b) being separated from each other (by 38a, 38b).

3. A single screw compressor according to claim 1, **characterised in that** each outlet (13a, 13b) communicated with respective unloading valve gear slidable in a respective recess (37a, 37b) opening from the bore (30) of the single screw.

4. A single screw compressor according to claim 1 **characterised in that** the pressure (Pe) of fluid fed to the secondary inlet of said one pumping chamber (10B) is in the range 0.3Pd to 0.7Pd, where Pd is the pressure at which fluid is discharged from said one pumping chamber.

5. A single screw compressor according to claim 4 **characterised in that** the pressure (Pe) is in the range 0.5Pd to 0.65Pd.

6. A single screw compressor according to claim 1, **characterised in that** means for liquid injection cooling is provided into at least one of the chambers (10A, 10B).

7. A single screw refrigeration compressor having two pumping chambers (10A, 10B) formed in the flutes of the screw and having an economiser port (12b) in at least one pumping chamber (10B) **characterised in that** means is provided to connect the outlet (13a) of the other chamber (10A) to the economiser port (12b) of the said one chamber (10B) to supercharge the compressor and improve performance without the need for any economiser vessel or side load.

8. A single screw compressor according to claim 7, **characterised in that** cooling and sealing of the compression process in at least one pumping chamber is provided by means permitting the injection of liquid refrigeration into said at least one chamber.

9. A single screw compressor according to claim 7, **characterised in that** the pressure of fluid fed to the economiser port (12b) is in the range 30% to 70% of the discharge pressure from the supercharged pumping chamber (10B).

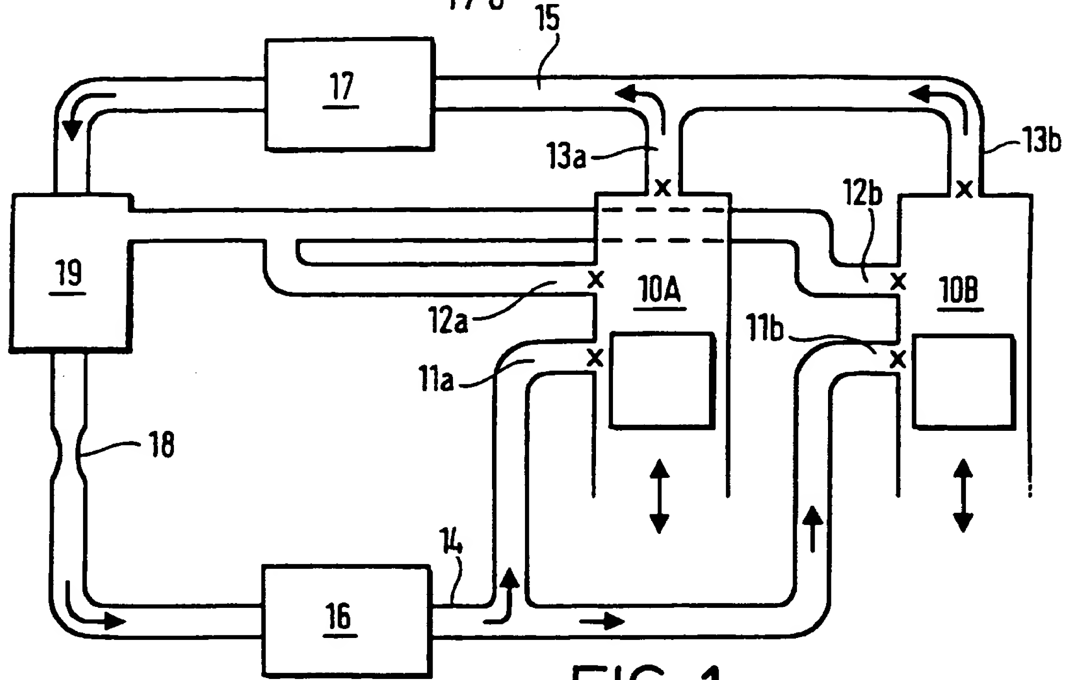
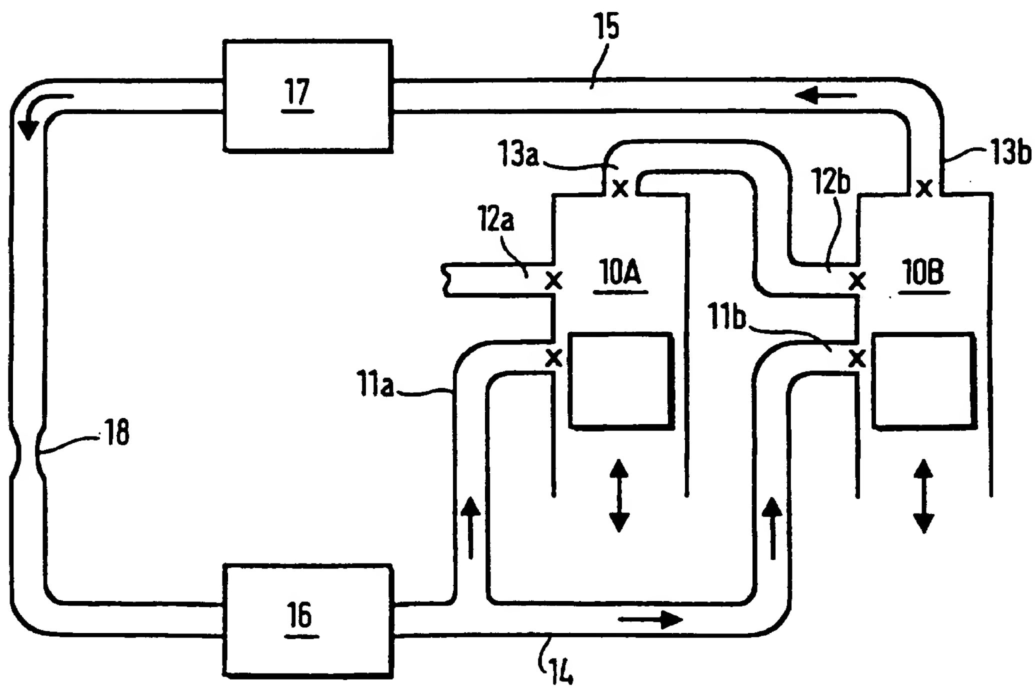

15
17
13a
13b
12b
19
10A
10B
12a
11b
11a
18
14
16

FIG. 1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13a
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12a
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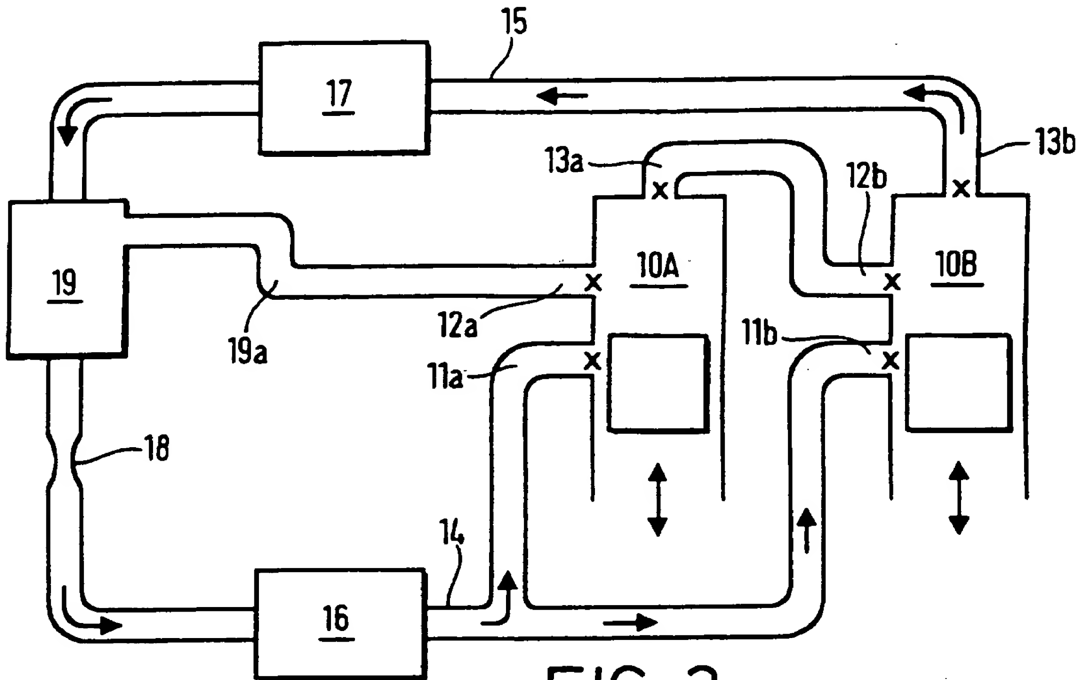
11b
11a
18
16
14

FIG. 2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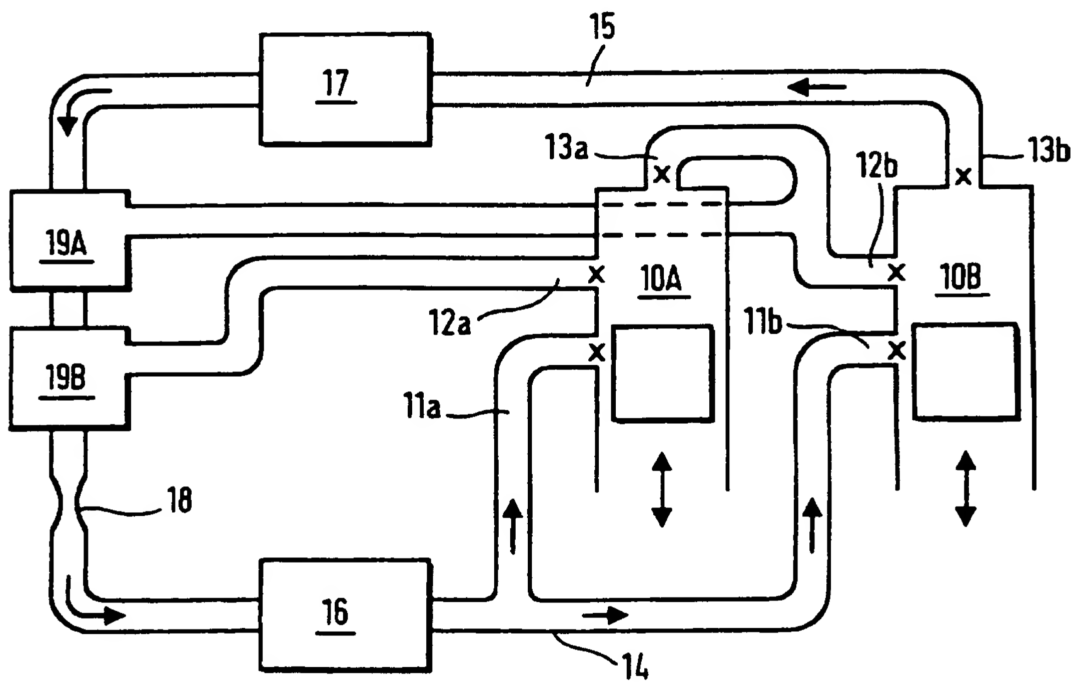
13b
19
10A
10B
19a
12a
11b
11a
18
14
16

FIG. 3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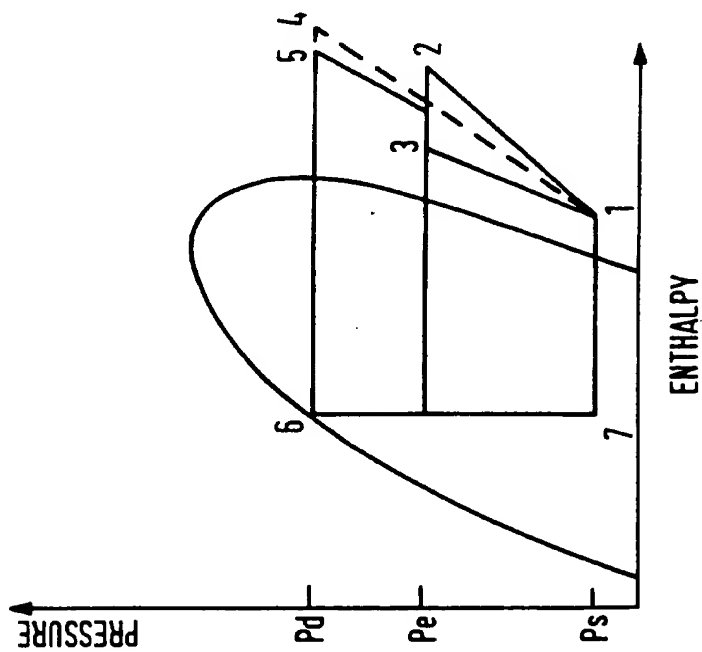
13a
12b
13b
19A
10A
10B
12a
11b
19B
11a
18
16
14

FIG. 4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Pe
Ps
ENTHALPY
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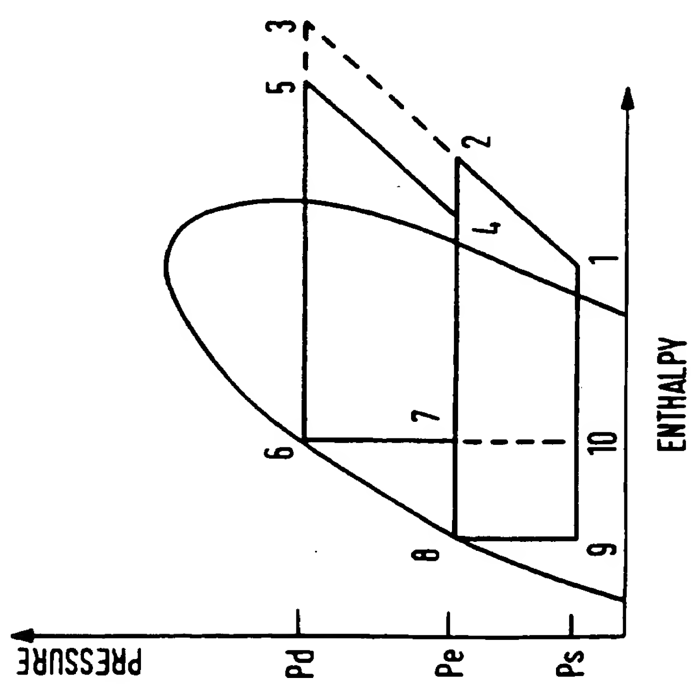
3
4
5
6
7

FIG. 6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ENTHALPY
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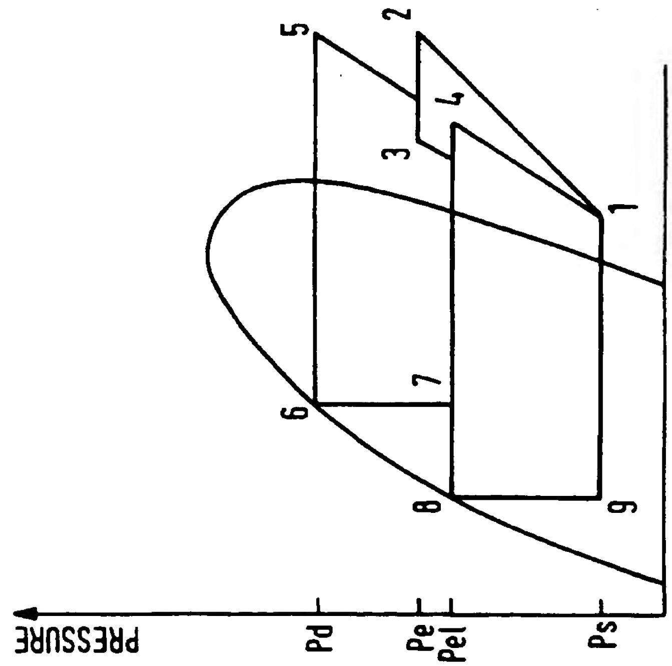
10

FIG. 5


PRESSURE
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Pel
Ps
ENTHALPY
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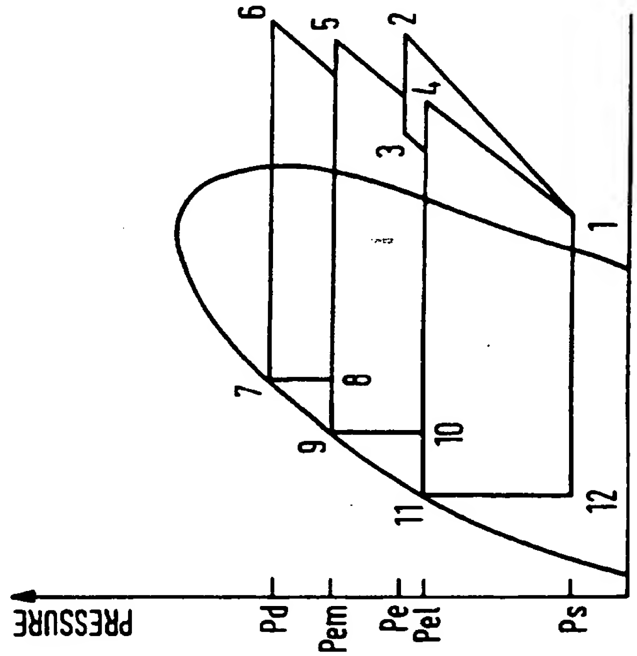
9

FIG. 7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Pe
Pel
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FIG. 8

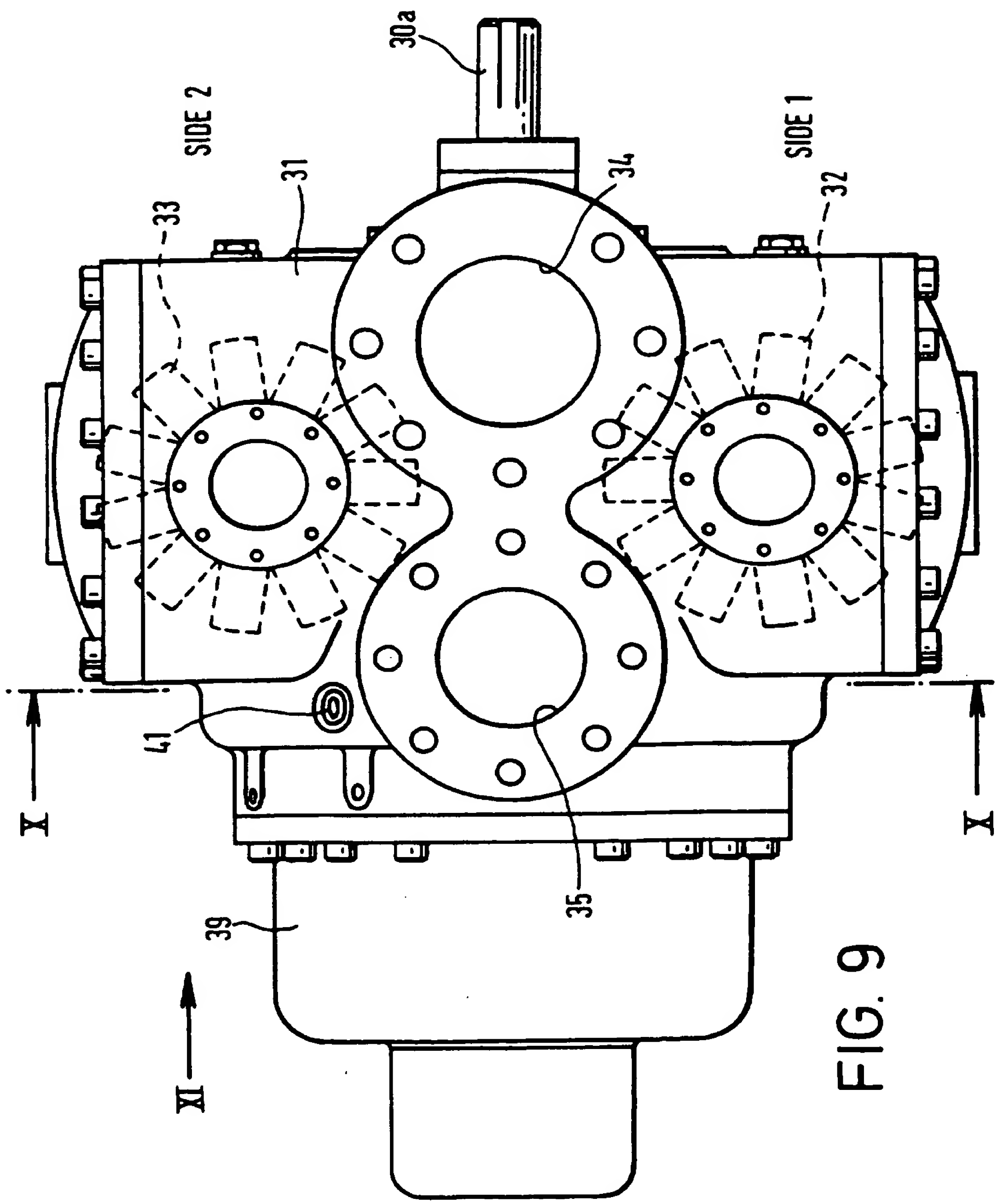

FIG. 9

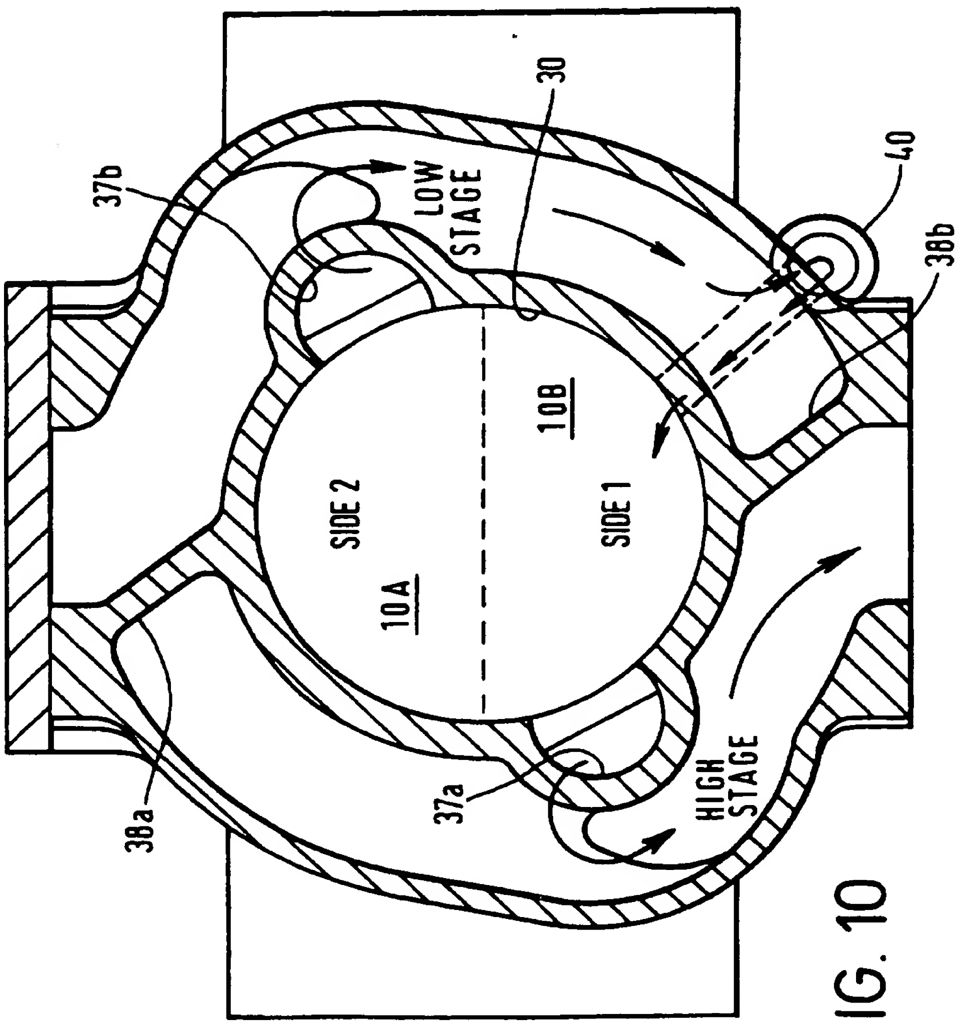

FIG. 10

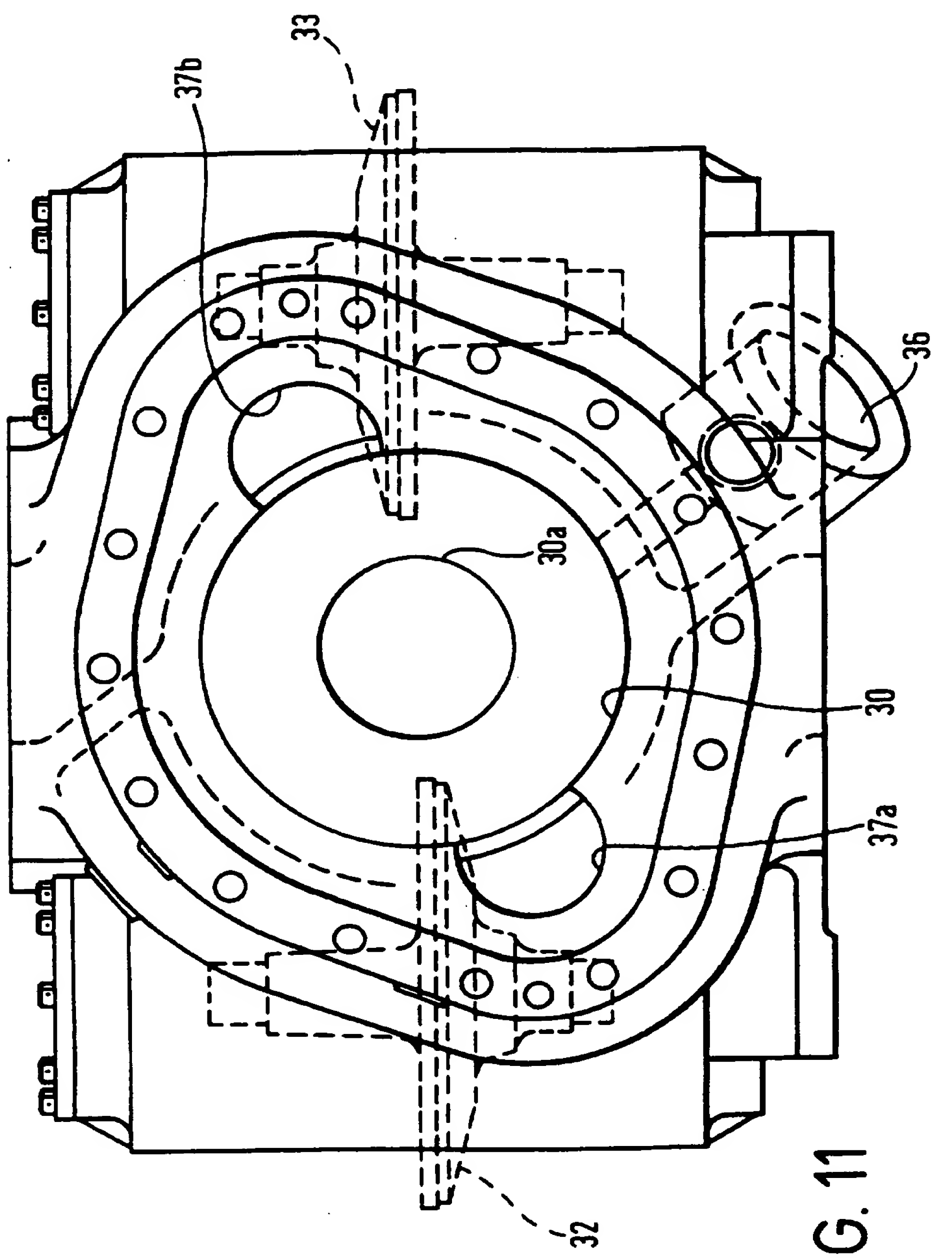

FIG. 11

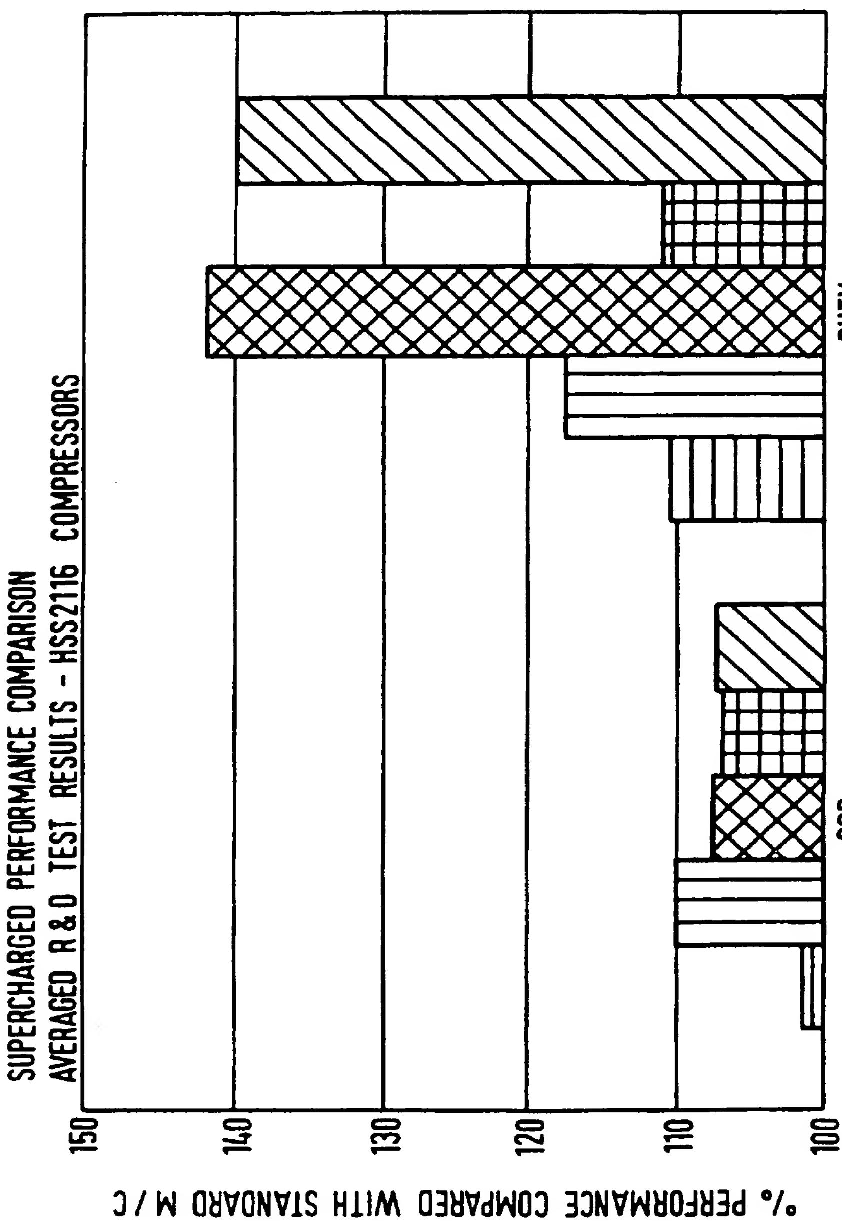
SUPERCHARGED PERFORMANCE COMPARISON
AVERAGED R & D TEST RESULTS - HSS2116 COMPRESSORS
150
140
130
120
110
100
% PERFORMANCE COMPARED WITH STANDARD M/C
COP
DUTY

FIG. 12

INTERNATIONAL SEARCH REPORT

Intern. ...al Application No
PCT/GB 96/02678

A. CLASSIFICATION OF SUBJECT MATTER
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According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 6 F04C

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	GB 1 501 474 A (UNISCREW LIMITED) 15 February 1978	1-5
Y	see the whole document ---	6-9
Y	US 4 696 627 A (ASANO ET AL.) 29 September 1987 see the whole document -----	6-9

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

* Special categories of cited documents :

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Date of the actual completion of the international search	Date of mailing of the international search report
21 February 1997	06.03.97

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Dimitroulas, P

INTERNATIONAL SEARCH REPORT

Information on patent family members

Inter nal Application No
PCT/GB 96/02678

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
GB-A-1501474	15-02-78	NONE	
US-A-4696627	29-09-87	JP-A- 62038888	19-02-87
		JP-A- 62041990	23-02-87